TITLE OF THE INVENTION

Hydrodynamic bearing and disc rotation apparatus using the same

BACKGROUND ART

1. TECHNICAL FIELD

The present invention relates to a hydrodynamic bearing having fluid in its rotation section and a disc rotation apparatus having the same.

2. PRIOR ART

In recent years, in recording apparatuses using discs and the like, their memory capacities are increasing and their data transfer speeds are rising. Hence, a disc rotation apparatus for use in this kind of recording apparatus is required to rotate at high speed and with high accuracy, and a hydrodynamic bearing is used in its rotating main shaft section.

A conventional hydrodynamic bearing and an example of a disc rotation apparatus having the same will be described below referring to FIG. 12. FIG. 12 is a cross-sectional view showing the right portion of the center of the rotation shaft, that is, the center line C, of the conventional hydrodynamic bearing. In FIG. 12, a shaft 31 is rotatably inserted into a sleeve 32 having a bearing hole 32A. At the lower end of the shaft 31, a flange 33 is provided so as to be

integrated therewith. The lower end of the flange 33 is accommodated in a recess portion formed by a hole in a base 35 and the sleeve 32 and rotatably held so as to be opposed to a thrust plate 34 mounted on the base 35. A hub rotor 36, a rotor magnet 38, a plurality of discs 39, spacers 40 and a clamper 41 are secured to the shaft 31. A motor stator 37 opposed to the rotor magnet 38 is installed on the base 35. Dynamic pressure generation grooves 32B and 32C indicated by broken lines are provided on the inner circumferential face of the bearing hole 32A of the sleeve 32. Dynamic pressure generation grooves 33A are provided on the upper face of the flange 33, a face opposed to the sleeve 32. In addition, dynamic pressure generation grooves 33B are provided on the lower face of the flange 33, a face opposed to the thrust plate 34. The clearances between the shaft 31 and the sleeve 32, including the dynamic pressure generation grooves 32B, 32C, 33A and 33B, are filled with oil.

The operation of the conventional hydrodynamic bearing shown in FIG. 12 will be described below. In FIG. 12, when electric power is applied to the coil of the stator 37, a rotating magnet field is generated, and a rotation force is generated in the rotor magnet 38, whereby the shaft 31

and the flange 33 rotate together with the hub rotor 36 and the discs 39. During the rotation, dynamic pressures are generated in the oil by the dynamic pressure generation grooves 32B, 32C, 33A and 33B, and the shaft 31 is floated in the upward direction of the figure and rotates while holding space from the sleeve 32 and without making contact with the thrust plate 34 and the sleeve 32. Magnet heads, not shown, make contact with the discs 39 and carry out the recording and reproduction of electrical signals.

The conventional hydrodynamic bearing configured as described above had problems described below. FIG. 13 is a plan view of the flange 33 which is provided with a plurality of the dynamic pressure generation grooves 33A indicated by black-colored regions. FIG. 14 is a bottom view of the flange 33 which is similarly provided with a plurality of the dynamic pressure generation grooves 33B indicated by black-colored regions. The outside diameters of the patterns of the dynamic pressure generation grooves 33A and 33B on the top and bottom faces are represented by D1o and D2o, respectively, and their inside diameters are represented by D1i and D2i, respectively. The diameters D1m and D2m of the respective turn-back parts of the dynamic pressure generation grooves 33A and 33B are set at sufficiently

large values so that pumping pressures in the
directions indicated by arrows E and F and by arrows G
and H, respectively, are raised.

FIG. 15 and FIG. 16 are views showing the
cross sections of relevant parts in the vicinity of
the lower end of the shaft 31 and showing pressures on
the surfaces of the flange 33 and the shaft 31 of the
above-mentioned conventional hydrodynamic bearing. If
the pumping pressures in the directions indicated by
arrows E and H shown in FIG. 13 and FIG. 14,
respectively, are raised too high, a negative pressure
with respect to atmospheric pressure is generated at
the central portion of the lower face of the flange 33
as indicated by curve P1 in FIG. 15, whereby air
bubbles mixed in the oil are coagulated and air is
accumulated in a region 43B having a constant size.

In FIG. 16, the dynamic pressure generation
grooves 32B and 32C of the sleeve 32 are made so that
dimension L1 in the figure is larger than dimension L2,
(L1 > L2), and so that dimension L4 is larger than
dimension L3, (L4 > L3). In addition, the dimensional
difference (L1 - L2) is selected so as to be nearly
equal to the dimensional difference (L4 - L3), that is,
(L1 - L2) $\fallingdotseq$ (L4 - L3). As shown by $\triangle$L in FIG. 16, in
the case that the amount of the oil becomes slightly
insufficient and the upper face of the oil is at the

position lower than the upper ends of the dynamic
pressure generation grooves 33B by dimension 4L, no
oil is present in the portion corresponding to the
dimension $\triangle L$ of the upper ends of the dynamic
pressure generation grooves 33B, whereby the pressure
distribution of oil is represented by curve P2 shown
in FIG. 16. In addition, a negative pressure is
generated at the lower portion of the range of the
dimension L4 in the figure. Hence, air bubbles are
accumulated in a region 43A, whereby there is a fear
of breaking the oil film in this region 43A and of
causing friction between the shaft 31 and the sleeve
32.

SUMMARY OF THE INVENTION

The present invention purports to provide a
hydrodynamic bearing in which a negative pressure is
prevented from generating between the shaft and the
sleeve, whereby oil film breakage due to locally
accumulated air in oil does not occur.

A hydrodynamic bearing in accordance with
the present invention comprises a sleeve having a
bearing hole at the nearly central portion thereof, a
shaft rotatably inserted into the bearing hole of the
above-mentioned sleeve, and a nearly disc-shaped
flange, secured to one end of the above-mentioned

shaft, one face of which is opposed to the end face of
the sleeve 1 and the other face of which is opposed to
a thrust plate provided to hermetically seal a region
including the above-mentioned end face of the above-
mentioned sleeve, wherein herringbone-shaped first and
second dynamic pressure generation grooves are
provided on at least one of the inner circumferential
face of the above-mentioned sleeve and the outer
circumferential face of the above-mentioned shaft so
as to be arranged in the direction along the shaft,
herringbone-shaped third dynamic pressure generation
grooves are provided on at least one of the opposed
faces of the flange and the thrust plate, the above-
mentioned first, second and third dynamic pressure
generation grooves are filled with oil having a
kinematic viscosity of 4 cSt (centi-stokes) or more at
40°C of temperature, one of the above-mentioned sleeve
and the above-mentioned shaft is secured to a base and
the other is secured to a rotatable hub rotor, and
when the outside diameter of the herringbone pattern
of the above-mentioned third dynamic pressure
generation groove is designated as d1o, the inside
diameter thereof is designated as d1i, the diameter of
the turn-back part thereof is designated as d1m, and
the diameter of the turn-back part of the herring
pattern, wherein the oil pressure generated by the

above-mentioned third dynamic pressure generation grooves in the direction from the outer circumference to the inner circumference of the flange becomes equal to the oil pressure generated in the direction from the inner circumference to the outer circumference thereof, is designated as dsy, the diameter d1m of the turn-back part is determined so that the following two equations are satisfied respectively:

$$d1m = dsy - (dsy - d1i) \times A$$
$$dsy = \{(d1i^2 + d1o^2)/2\}^{1/2},$$

where A is a value in the range of 0.05 or more to less than 1.0.

A hydrodynamic bearing in accordance with another aspect of the present invention comprises a sleeve having a bearing hole at the nearly central portion thereof, a shaft rotatably inserted into the bearing hole of the above-mentioned sleeve, and a nearly disc-shaped flange, secured to one end of the above-mentioned shaft, one face of which is opposed to the end face of the sleeve 1 and the other face of which is opposed to a thrust plate provided to hermetically seal a region including the above-mentioned end face of the above-mentioned sleeve, wherein herringbone-shaped first and second dynamic pressure generation grooves are provided on at least one of the inner circumferential face of the above-

mentioned sleeve and the outer circumferential face of the above-mentioned shaft so as to be arranged in the direction along the shaft, among the above-mentioned first and second dynamic pressure generation grooves, when the grooves away from the above-mentioned thrust plate are designated as the first dynamic pressure generation grooves and the grooves close thereto are designated as the second dynamic pressure generation grooves, the first length L1 of the groove portion, away from the above-mentioned thrust plate, of the above-mentioned herringbone-shaped first dynamic pressure generation groove in the direction of the shaft is larger than the second length L2 of the groove portion close to the above-mentioned thrust plate in the direction of the shaft, the above-mentioned herringbone-shaped second dynamic pressure generation groove is made symmetric with respect to a line passing through the herringbone-shaped turn-back parts, the value of a calculation expression, (L1 + L2)/(2 X L2) represented by using the above-mentioned first length L1 and the above-mentioned second length L2, is in the range of 1.02 to 1.60, herringbone-shaped third dynamic pressure generation grooves are provided on at least one of the opposed faces of the flange and the thrust plate, the above-mentioned first, second and third dynamic pressure generation grooves

are filled with oil having a kinematic viscosity of 4
cSt or more at 40°C of temperature, and one of the
above-mentioned sleeve and the above-mentioned shaft
is secured to a base and the other is secured to a
rotatable hub rotor.

A hydrodynamic bearing in accordance with
another aspect of the present invention comprises a
sleeve having a bearing hole at the nearly central
portion thereof, a shaft rotatably inserted into the
bearing hole of the above-mentioned sleeve, and a
nearly disc-shaped flange, secured to one end of the
above-mentioned shaft, one face of which is opposed to
the end face of the sleeve 1 and the other face of
which is opposed to a thrust plate provided to
hermetically seal a region including the above-
mentioned end face of the above-mentioned sleeve,
wherein herringbone-shaped first and second dynamic
pressure generation grooves are provided on at least
one of the inner circumferential face of the above-
mentioned sleeve and the outer circumferential face of
the above-mentioned shaft, among the above-mentioned
first and second dynamic pressure generation grooves,
when the grooves away from the above-mentioned thrust
plate are designated as the first dynamic pressure
generation grooves and the grooves close thereto are
designated as the second dynamic pressure generation

grooves, the first length L1 of the groove portion, away from the above-mentioned thrust plate, of the above-mentioned herringbone-shaped first dynamic pressure generation groove in the direction of the shaft is larger than the second length L2 of the groove portion close to the above-mentioned thrust plate in the direction of the shaft, the above-mentioned herringbone-shaped second dynamic pressure generation groove is made symmetric with respect to a line passing through the herringbone-shaped turn-back parts, the value of a calculation expression, (L1 + L2)/(2 X L2) represented by using the above-mentioned first length L1 and the above-mentioned second length L2, is in the range of 1.02 to 1.60, herringbone-shaped third dynamic pressure generation grooves are provided on at least one of the opposed faces of the flange and the thrust plate, the above-mentioned first, second and third dynamic pressure generation grooves are supplied with oil having a kinematic viscosity of 4 cSt or more at 40° C of temperature, one of the above-mentioned sleeve and the above-mentioned shaft is secured to a base and the other is secured to a rotatable hub rotor, and when the outside diameter of the herringbone pattern of the above-mentioned third dynamic pressure generation groove is designated as dlo, the inside diameter thereof is designated as dli,

the diameter of the turn-back part thereof is designated as dlm, and the diameter of the turn-back part of the herring pattern, wherein the oil pressure generated by the above-mentioned third dynamic pressure generation grooves in the direction from the outer circumference to the inner circumference of the flange becomes equal to the oil pressure generated in the direction from the inner circumference to the outer circumference thereof, is designated as dsy, the diameter dlm of the turn-back part is determined so that the following equations are satisfied respectively:

$$dlm = dsy - (dsy - dli) \times A$$
$$dsy = \{(dli^2 + dlo^2)/2\}^{1/2},$$

where A is a value in the range of 0.05 or more to less than 1.0.

In accordance with the above-mentioned configurations of the present invention, the patterns of the dynamic pressure generation grooves in the thrust bearing section and the radial bearing section have optimum shapes, whereby no negative pressure is generated inside the bearing. Hence, since air accumulation due to the coagulation of air bubbles can be prevented, it is possible to provide a hydrodynamic bearing not causing oil film breakage.

A disc rotation apparatus using the

hydrodynamic bearing in accordance with the present invention records or reproduces signals, wherein a recording/reproduction disc is concentrically secured to the hub rotor of the hydrodynamic bearing in accordance with claims 1 to 5 and rotated, magnetic heads or optical heads are provided so as to be opposed to the faces of the above-mentioned rotating disc, and the magnetic heads or optical heads are configured so as to be movable in parallel with the faces of the above-mentioned disc. By using the hydrodynamic bearing in accordance with the present invention, it is possible to obtain a disc rotation apparatus being high in the reliability of the bearing.

BRIEF DESCRIPTION OF THE DRAWINGS

FIG. 1 is a cross-sectional view of a hydrodynamic bearing in accordance with a preferred embodiment of the present invention;

FIG. 2 is a bottom view of the flange 3 of the hydrodynamic bearing in accordance with this embodiment;

FIG. 3 is a plan view of the flange 3 of the hydrodynamic bearing in accordance with this embodiment;

FIG. 4 is a graph showing the relationship between the pump pressure in the dynamic pressure

generation grooves 3A of the flange 3 and the

dimensional distribution of the diameter dsy of the

turn-back part and the inside diameter dli of the

dynamic pressure generation groove 3A in the

hydrodynamic bearing in accordance with this

embodiment;

FIG. 5 is a cross-sectional view of a

relevant part showing the distribution of oil pressure

generated by the dynamic pressure generation grooves

3A and 3B in the case when the floating distance S1

between the flange 3 and the thrust plate 4 is

sufficiently small in the hydrodynamic bearing in

accordance with this embodiment;

FIG. 6 is a cross-sectional view of a

relevant part showing the distribution of oil pressure

generated by the dynamic pressure generation grooves

3A and 3B in the case when the floating distance S2

between the flange 3 and the thrust plate 4 is

sufficiently large in the hydrodynamic bearing in

accordance with this embodiment;

FIG. 7 is a cross-sectional view of a

relevant part showing the distribution of oil pressure

in the radial bearing section and the distribution of

oil pressure generated by the dynamic pressure

generation grooves 3A and 3B of the flange 3 of the

hydrodynamic bearing in accordance with this

embodiment;

FIG. 8 is a cross-sectional view of a relevant part showing the distribution of oil pressure in the radial bearing section and the distribution of oil pressure generated by the dynamic pressure generation grooves 3A and 3B of the flange 3 in the case when the amount of oil is smaller than a specified amount in the hydrodynamic bearing in accordance with this embodiment;

FIG. 9 is a graph showing the relationship between the oil pressure generated by the dynamic pressure generation grooves 1A and 1B and the dimensional distribution of the dynamic pressure generation grooves 1A and 1B of the hydrodynamic bearing in accordance with this embodiment;

FIG. 10 is a graph showing the relationship between the bubble entry amount in oil and the kinematic viscosity of oil at 40°C of temperature in the hydrodynamic bearing in accordance with this embodiment;

FIG. 11 is a cross-sectional view of a disc rotation apparatus using the hydrodynamic bearing in accordance with this embodiment of the present invention;

FIG. 12 is the cross-sectional view showing the right half of the conventional hydrodynamic

bearing;

FIG. 13 is the plan view of the flange 33 of
the conventional hydrodynamic bearing;

FIG. 14 is the bottom view of the flange 33
of the conventional hydrodynamic bearing;

FIG. 15 is the cross-sectional view of a
relevant part showing the distribution of oil pressure
generated by the dynamic pressure generation grooves
33A and 33B of the flange 33 in the conventional
hydrodynamic bearing; and

FIG. 16 is the cross-sectional view of a
relevant part showing the distribution of oil pressure
generated in the radial direction by the dynamic
pressure generation grooves 32B and 3C of the sleeve
32 of the conventional hydrodynamic bearing.

DESCRIPTION OF PREFERRED EMBODIMENT

A preferred embodiment of a hydrodynamic
bearing in accordance with the present invention will
be described below referring to FIGS. 1 to 10. FIG. 1
is a cross-sectional view of a hydrodynamic bearing in
accordance with an embodiment of the present invention.
In FIG. 1, a sleeve 1 has a bearing hole 20 at its
nearly central portion, and herringbone-shaped dynamic
pressure generation grooves 1A and 1B are formed on
the inner circumferential face of the bearing hole 20.

A recess portion 1C is formed at the lower end of the sleeve 1. A shaft 2 is rotatably inserted into the bearing hole 20. A flange 3 is secured to the lower end of the shaft 2 so as to be accommodated in the recess portion 1C at the lower end of the sleeve 1. A thrust plate 4 is secured to the recess portion 1C of the sleeve 1 by a securing method, such as laser welding, precision crimping or bonding, and the recess portion 1C including the flange 3 is hermetically sealed. The sleeve 1 is secured to a base 6. The shaft 2 is secured to a hub rotor 7. Dynamic pressure generation grooves are provided on one of the opposed faces of the flange 3 and the thrust plate 4. In FIG. 1, dynamic pressure generation grooves 3A are provided on the lower face of the flange 3. Dynamic pressure generation grooves 3B are also provided on the upper face of the flange 3 opposed to the recess portion 1C of the sleeve 1. The insides of the dynamic pressure generation grooves 1A, 2A(sic)--1B--, 3A and 3B are filled with oil or grease. A rotor magnet 9 is installed in the hub rotor 7. In addition, a stator 8 is installed on the base 6 so as to be opposed to the above-mentioned rotor magnet 9. Two discs 10, for example, are installed on the hub rotor 7 via a spacer 12. The discs 10 are secured by a clamper 11 installed on the shaft 2 by a screw 13.

The operation of the hydrodynamic bearing in accordance with this embodiment configured as mentioned above will be described by with reference to FIGS. 1 to 10. In FIG. 1, first, when electric power is applied to the coil of the stator 8, a rotating magnet field is generated, and the rotor magnet 9 receives a rotation force, and the hub rotor 7, the shaft 2 and the discs 10 rotate together with the clamper 11 and the spacer 12. By the rotation, the dynamic pressure generation grooves 1A, 1B, 3A and 3B rake up oil, and pressures are generated between the dynamic pressure generation grooves 1A and 1B and the shaft 2 and between the dynamic pressure generation grooves 3A and the thrust plate 4. Hence, the shaft 2 is floated in the upward direction of the figure and rotates without making contact with the thrust plate 4 and the sleeve 1.

FIG. 2 is a view of the lower face of the flange 3, that is, the bottom face thereof opposing to the thrust plate 4, and the black-colored portions indicate the dynamic pressure generation grooves 3A. The outside diameter of the pattern of the dynamic pressure generation groove 3A is designated as d1o, the inside diameter thereof is designated as d1i and the diameter of the turn-back part is designated as d1m. When the flange 3 rotates inside the recess

portion 1C of the sleeve 1, an oil pressure G is
generated on the face of the flange 3 in the direction
from the outer circumference to the inner
circumference thereof. Furthermore, an oil pressure H
is also generated in the direction from the inner
circumference to the outer circumference thereof. The
diameter of the turn-back part wherein the pressure G
becomes equal to the pressure H is represented by dsy.
Usually, the dynamic pressure generation grooves 3A
are designed so that the pressure G becomes equal to
the pressure H. For this purpose, the diameter d1m is
determined by equation (1), a well-known equation in
hydrodynamics.

$$d1m = \{(d1i^2 + d1o^2)/2\}^{1/2} \ldots (1)$$

However, the hydrodynamic bearing in
accordance with the present invention is designed so
that the pressure G becomes larger than the pressure H.
This is represented by equations (2) and (3).

$$d1m = dsy - (dsy - d1i) \times Y \ldots (2)$$

$$dsy = \{(d1i^2 + d1o^2)/2\}^{1/2} \ldots (3)$$

In equation (2), Y is designed so as to be
in the range of 0.05 to 1.0.

FIG. 3 is a plan view of the flange 3, and
the black-colored portions indicate the dynamic
pressure generation grooves 3B. The dynamic pressure
generation grooves 3B are designed so that the

pressure in the direction indicated by arrow E from
the inner circumference to the outer circumference is
nearly balanced with the pressure in the direction
indicated by arrow F from the outer circumference to
the inner circumference. In other words, when the
outside diameter of the pattern of the dynamic
pressure generation groove 3B is designated as d2o,
the inside diameter thereof is designated as d2i and
the diameter of the turn-back part thereof is
designated as d2m, a relationship represented by
equation (4) is established.

$$d2m = \{(d2o^2 + d2i^2)/2\}^{1/2} \ldots (4)$$

The vertical axis of the graph in FIG. 4
represents an oil pressure (pascal) in the dynamic
pressure generation groove 3A, which is variable
depending on the value of the above-mentioned Y. The
horizontal axis represents the value of equation (dsy
- dlm)/(dsy - dli). If asymmetry is insufficient in
the pressures inside the bearing, a partially negative
pressure portion is generated somewhere inside the
bearing, and air may be accumulated there. On the
other hand, if asymmetry is excessive, the internal
pressure becomes too high, and there arises a danger
of causing cavitation or microbubbles. Relating to
the hydrodynamic bearing in accordance with this
embodiment, a hydrodynamic bearing is made by using

transparent materials for the sake of observation, and experiments are carried out. As a result, it was found that when the value of the above-mentioned Y was in the range of 0.05 to 1.0, the amount of air bubbles entered and the amount of air accumulated during rotation were minimal, whereby this range was an appropriate range and air was most hardly accumulated in oil.

FIG. 5 is a cross-sectional view showing the cross-section of a relevant part and the pressure distribution of oil by the dynamic pressure generation grooves 3A and 3B with reference to the atmospheric pressure in the case that the floating amount (S1) of the flange 3 from the thrust plate 4 is sufficiently small. In the hydrodynamic bearing in accordance with the present invention, only the positive pressure indicated by curve P10 representing the pressure distribution of oil is generated and no negative pressure is generated. For this reason, a phenomenon of air accumulation between the flange 3 and the thrust plate 4 hardly occurs.

FIG. 6 is a cross-sectional view showing the cross-section of a relevant part and the pressure distribution of oil by the dynamic pressure generation grooves 3A and 3B as indicated by pressure curves P11 and P12 in the case that the floating amount (S2) is

sufficiently large. Even in this case, no negative
pressure is generated inside the bearing as indicated
by the pressure curve P11. In FIG. 6, the positive
pressure indicated by the curve P12 of the pressure
generated by the dynamic pressure generation grooves
3B on the upper face of the flange 3 prevents
collision between the flange 3 and the sleeve 1.

FIG. 7 and FIG. 8, views showing the cross-
sections of a relevant part and the pressure
distributions, show detailed pressure distributions
regarding the pressures generated in the radial
direction (the left-to-right direction in the figure)
of the dynamic pressure generation grooves 1A and 1B.
FIG. 7 shows a case wherein the clearance portions of
the hydrodynamic bearing are wholly filled with oil 5
and the liquid face is above the upper ends of the
dynamic pressure generation grooves 1A. The dynamic
pressure generation grooves 1A are provided in the
upper portion of the sleeve 1 and made asymmetric such
that the groove portion 28A in the range of the upper
half dimension L1 is longer than the groove portion
29A in the range of the lower half dimension L2.
Hence, the oil is pressed downward by the effect of
dynamic pressure, thereby being prevented from leaking
outside. The acute connection part of the groove
portion 28A and the groove portion 29A is referred to

as a turn-back part. The groove portion 28A and the groove portion 29A of the dynamic pressure generation groove 1A have the same inclination angle. In the configuration shown in FIG. 7, if the difference between the dimension L1 and the dimension L2 of the dynamic pressure generation groove 1A is too small, there is a danger of causing oil leakage. On the other hand, if the difference is too large, the internal pressure becomes too high, and there is a danger of generating cavitation or microbubbles.

In the dynamic pressure generation groove 1B, the groove portion 28B of the upper half is made symmetric with the groove portion 29B of the lower half. Since the dynamic pressure generation groove 1A is made asymmetric, the pressure inside the bearing becomes positive as indicated by pressure curve P13. Since no negative pressure is generated inside the bearing even in this case, air accumulation hardly occurs. The pressures in the thrust direction become positive as indicated by pressure curves P14 and P15, whereby no negative pressure is generated.

FIG. 8 shows a case wherein the oil inside the bearing decreases and becomes insufficient by the amount corresponding to the dimension ΔL. Even in this case, only the positive pressure is generated as indicated by pressure curve P17, whereby no negative

pressure is generated inside the bearing.

FIG. 9 shows the appropriate range of the asymmetry of the dynamic pressure generation groove 1A. It is desirable that the dimension L2 of the groove portion 29A is smaller than the dimension L1 of the groove portion 28A, that is, the portion on the opposite side thereof, and that the value of the relational expression shown on the left side of equation (5) with respect to the dimensions L1 and L2 is in the range of values shown on the right side.

$$(L1 + L2)/(2 \times L2) = 1.02 \text{ to } 1.60 \ldots (5)$$

In the range indicated in equation 5, the entry of air and the entry of microbubbles hardly occurred.

FIG. 10 shows the relationship between the kinematic viscosity of oil or the kinematic viscosity of the base oil of grease and the bubble mixing rate into the clearances of the bearing, obtained from the observation results of the experimental bearing made of the transparent materials. The bubble mixing rate is represented by the percentage of the volume of bubbles with respect to the volume of oil. According to the observation results, it was found that the bubble mixing rate was very low in the case when oil or the base oil of grease had a kinematic viscosity of 4 cSt or more at 40° C of temperature.

The configuration and operation of a disc
rotation apparatus using the hydrodynamic bearing in
accordance with the present invention will be
described by using FIG. 11. In FIG. 11, on a
hydrodynamic bearing provided inside a box-shaped base
6 and comprising a sleeve 1, a shaft 2, a flange 3, a
thrust plate 4, a hub rotor 7, a stator 8 and a rotor
magnet 9, two discs 10 are installed while space is
provided therebetween by using a spacer 12. Heads 25
respectively supported by arms 15 are opposed to both
faces of the disc 10. The arms 15 rotate while being
supported by a head support shaft 16. The upper face
of the base 6 is hermetically sealed by an upper lid
14 so as to prevent the entry of dust and the like.
When electric power is applied to the motor stator 8,
a rotating magnet field is generated, and the rotor
magnet 9 starts rotating together with the hub rotor 7,
the shaft 2 and the discs 10. The dynamic pressure
generation grooves 1A, 1B, 3A and 3B rake up oil by
pumping forces and generate pressures, whereby the
bearing portion floats and rotates with high accuracy
in a noncontact state. The heads 25 make contact with
the rotating discs 10, thereby recording or
reproducing electrical signals.

Although the thrust plate is secured to the
sleeve 1 in FIG. 1, it may be secured to the base 6 if

the interior of the bearing can be hermetically sealed.

Even if helical dynamic pressure generation grooves, in which d1m = d1o, are used as a modification application example of the dynamic pressure generation grooves 3A shown in FIG. 2, instead of the herringbone-shaped grooves, nearly equivalent performance can be obtained.

As mentioned above, with the hydrodynamic bearing in accordance with this embodiment, the entry of air into the hydrodynamic bearing section is prevented, and the breakage of oil film, having been apt to occur in bearings, is prevented. As a result, a long-life disc rotation apparatus capable of rotating discs with high accuracy is obtained by using the hydrodynamic bearing in accordance with the present invention.

In addition, the design conditions of the dynamic pressure generation grooves are combined with the selection conditions of the kinematic viscosity of oil so that the accumulation of air inside the bearing due to the pumping forces in the dynamic pressure generation grooves is prevented during rotation, therefore the breaking of oil film in the clearances of the bearing does not occur, whereby the hydrodynamic bearing in accordance with the present invention has high accuracy and long life.